# NASA TECHNICAL NOTE

NASA TN D-5800

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# PERFORMANCE OF 75-MILLIMETER-BORE BEARINGS TO 1.8 MILLION DN WITH ELECTRON-BEAM-WELDED HOLLOW BALLS

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MAY 1970

ERRATA

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The following list should appear as page 22:

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NASA-Langley, 1970

Issued 10-12-70

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1. Report No.
NASA TN D-5800

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle PERFORMANCE OF 75-MILLIMETER-BORE BEARINGS TO 1.8 MILLION DN WITH ELECTRON-BEAM-WELDED HOLLOW BALLS

5. Report Date
May 1970

6. Performing Organization Code

7. Author(s) Harold H. Coe, Herbert W. Scibbe, and Richard J. Parker

8. Performing Organization Report No.
E-5442

9. Performing Organization Name and Address
Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio 44135

10. Work Unit No.
126-15

11. Contract or Grant No.

12. Sponsoring Agency Name and Address
National Aeronautics and Space Administration
Washington, D.C. 20546

13. Type of Report and Period Covered
Technical Note

14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

An experimental investigation was performed with two series (115 and 215) of 75-mm-bore ball bearings using hollow balls as the rolling elements. Similar bearings using solid balls were also tested and the data compared. The bearings were tested at 500- and 1000-lb (2200- and 4400-N) thrust loads at shaft speeds up to 24 000 rpm ($1.8\times10^6$ DN). The results showed only slight differences in torque and outer-race temperature between the hollow ball bearings and the corresponding solid ball bearings. The 11/16-in.- (17.5-mm-) diameter AISI M-50 steel hollow balls, for use with the series 215 bearings, were tested in a five-ball fatigue tester. The resulting fatigue life was significantly less than expected. The balls failed in flexure fatigue because of a stress concentration in the weld area.

17. Key Words (*Suggested by Author(s)*)
Bearing
Hollow balls
Electron-beam welded
Fatigue life

18. Distribution Statement
Unclassified - unlimited

| 19. Security Classif. (of this report) | 20. Security Classif. (of this page) | 21. No. of Pages | 22. Price * |
|---|---|---|---|
| Unclassified | Unclassified | 23 | $3.00 |

*For sale by the Clearinghouse for Federal Scientific and Technical Information
Springfield, Virginia 22151

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## SUMMARY

Experiments were performed to compare the operating characteristics of ball bearings using hollow balls as the rolling elements with similar bearings using solid balls.

Two series (115 and 215) of 75-millimeter-bore bearings were tested with thrust loads of 500 and 1000 pounds (2200 and 4400 N) at shaft speeds up to 24 000 rpm or $1.8\times10^{6}$ DN (bearing bore in mm times shaft speed in rpm). The series 115 bearings used 1/2-inch- (12.7-mm-) diameter SAE 52100 steel balls. The hollow balls used with these bearings had a 0.100-inch (2.5-mm) wall thickness (21-percent weight reduction). The series 215 bearings used 11/16-inch- (17.5-mm-) diameter balls, and the AISI M-50 steel hollow balls had a 0.060-inch (1.5-mm) wall thickness (56-percent weight reduction).

The results of the tests with the series 115 bearings showed very little difference in torque or outer-race temperature between the bearings with the hollow balls and those with the solid balls. The tests with the series 215 bearings showed that the torque and outer-race temperature of the bearings with the hollow balls were slightly lower than those with the solid balls over the range of test conditions.

The results from a previous investigation showed that rolling-element fatigue life of the 1/2-inch- (12.7-mm-) diameter hollow balls was not significantly different from that of solid balls of the same material. The fatigue life of the 11/16-inch- (17.5-mm-) diameter hollow balls, determined in the NASA five-ball fatigue tester at a maximum Hertz stress of 700 000 psi ($4.82\times10^{9}$ N/m$^{2}$), was significantly less than that of previously run solid balls. The 11/16-inch (17.5-mm) hollow balls in the fatigue tests and in the series 215 bearing tests failed by flexure fatigue. Cracks initiated at the inner diameter of the ball at a stress concentration in the weld area and propagated through the wall, usually terminating in a spall on the outer surface.

# INTRODUCTION

Recent developments in gas turbine engines, such as advanced compressor design, high-temperature materials, and increased power output, have resulted in a requirement for larger shaft diameters and higher main shaft bearing speeds (ref. 1). Bearings in current production aircraft turbine engines operate in the range from 1.5 to 2 million DN (bearing bore in mm times shaft speed in rpm). Engine designers anticipate that, during the next decade, turbine bearing DN values will have to increase to the range of 2.5 to 3 million. It is speculated that, after 1980, turbine engine developments may require bearing DN values as high as 4 million.

When ball bearings are operated at DN values above 1.5 million, centrifugal forces produced by the balls can become significant. The resulting increase in Hertz stresses at the outer-race-ball contacts can seriously affect bearing fatigue life (ref. 2).

Operation at high DN values also results in high heat generation. Excessive heat generation can result from ball skidding (ref. 3) or ball spinning (ref. 4). The heat generated within the bearing must be removed, or a loss in operating clearance will occur and the bearing will fail. Refinements in bearing internal geometry, such as the use of smaller diameter balls, larger diametral clearances, or more open race curvatures, can effectively reduce heat generation and/or centrifugal force. The degree of improvement is limited, however, since the geometrical changes that reduce heat generation will generally reduce fatigue life (ref. 5).

Another possible solution to problems caused by ball centrifugal force or excessive heat generation is the use of hollow balls in the bearing. Obviously, removal of a large percentage of the mass from the ball will reduce the centrifugal force. In reference 6, an analysis was made that predicted some advantages of hollow balls over solid balls in a high-speed thrust bearing. The analysis showed that when hollow balls with an outer- to inner-diameter ratio of 1.25 (51-percent weight reduction) were used at a DN value of 3 million, the ball spin-to-roll ratio was reduced by as much as 25 percent. (Ball spin-to-roll ratio is an indication of the total rolling and spinning torque. The lower this ratio, the lower the bearing ball-spin torque and, thus, the lower the heat generation.)

The feasibility of fabricating hollow balls by electron-beam welding two hemispherically formed shells has been adequately demonstrated (refs. 7 to 9). In reference 9, rolling-element fatigue tests were conducted with 1/2-inch- (12.7-mm-) diameter solid and hollow balls made from SAE 52100 consumable-electrode vacuum-melted steel. The hollow balls had a 0.100-inch (2.54-mm) wall thickness (21-percent weight reduction). The fatigue lives of these hollow balls compared favorably with those of the 1/2-inch (12.7-mm) solid balls fabricated from the same heat of material. From the limited number of hollow ball failures, it was concluded that the weld zone was not appreciably weak in fatigue. It was also determined that the weld zone had a hardness and microstructure similar to that of the parent material in the ball.

The objectives of the present investigation were to (1) determine experimentally the operating characteristics of a full-scale bearing using hollow balls as the rolling elements, (2) compare the torque and outer-race temperature data of a bearing with the hollow balls with data of a similar bearing with solid balls, and (3) determine the fatigue lives of hollow balls with a weight reduction of approximately 50 percent.

Bearing tests were conducted with two series (115 and 215 series) of 75-millimeter-bore deep-groove ball bearings using both hollow and solid balls. The bearings were operated at thrust loads of 500 and 1000 pounds (2200 and 4400 N) at speeds to 24 000 rpm ($1.8\times10^{6}$ DN) using air-oil-mist lubrication. The hollow balls for the series 115 bearings had a 1/2-inch- (12.7-mm-) diameter and a 0.100-inch (2.54-mm) wall thickness (a weight reduction of 21.7 percent from that of the solid balls). The hollow balls for the series 215 bearings had a 11/16-inch (17.5-mm) diameter and a 0.060-inch (1.5-mm) wall thickness (a weight reduction of 56.5 percent from that of the solid balls).

The NASA five-ball fatigue tester was used to determine the life of the 11/16-inch- (17.5-mm-) diameter hollow balls. Tests were conducted at a maximum Hertz stress of 700 000 psi ($4.82\times10^{9}$ $N/m^{2}$), a shaft speed of 10 600 rpm, and a contact angle of $34.5^{\circ}$.

## APPARATUS AND INSTRUMENTATION

### Bearing Test Rig

A cutaway view of the bearing test apparatus is shown in figure 1. A variable-speed, direct-current motor drives the test bearing shaft through a gear speed increaser. The ratio of the test shaft speed to the motor shaft speed was 14.

The test shaft was supported and cantilevered at the driven end by two oil-jet-lubricated ball bearings. The test bearing was thrust loaded by a pneumatic cylinder through an externally pressurized gas thrust bearing.

Bearing torque was measured with an unbonded strain-gage force transducer connected to the periphery of the test bearing housing, as shown in figure 1. This torque was recorded continuously by a millivolt potentiometer. The estimated accuracy of the data recording system was ±0.05 pound-inch (0.006 N-m).

Bearing outer-race temperature was measured with two iron-constantan thermocouples located as shown in figure 1. The estimated accuracy of the temperature measuring system was about $\pm2^{\circ}$ F (±1 K).

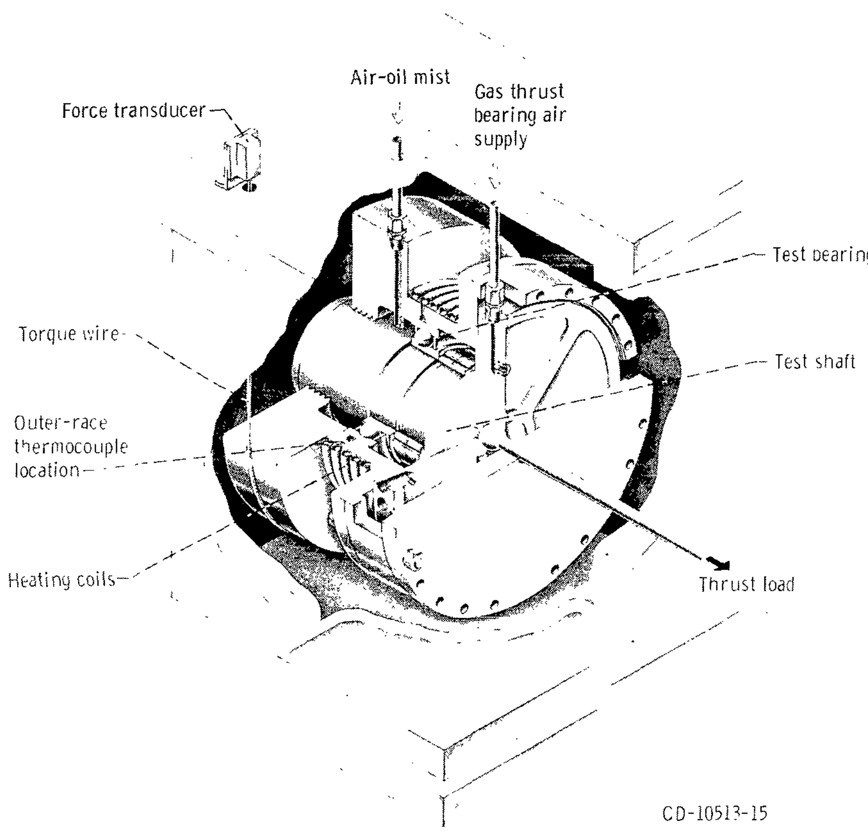



Figure 1. - Bearing test apparatus.

## Test Bearing Lubrication System

The test bearing was lubricated by an air-oil mist introduced to the bearing as shown in figure 1. The air-oil-mist generating system (fig. 2) was used previously for a minimum oil flow investigation (ref. 10). The pressurized tank fed oil through the capillary tube into the air line downstream from a Venturi. A jet of high-velocity air generated by the Venturi atomized a metered flow of oil from the capillary into an air-oil mist. The mist then flowed to the test bearing lubricator through plastic tubing. The oil flow rates ranged from 0.01 to 0.07 pound per minute ($0.8\times10^{-4}$ to $5.3\times10^{-4}$ kg/sec). The lubricant used in this investigation was a super-refined naphthenic mineral oil with a viscosity of 75 centistokes at $100^{\circ}$ F ($75\times10^{-6}$ $m^2$/sec at 311 K).

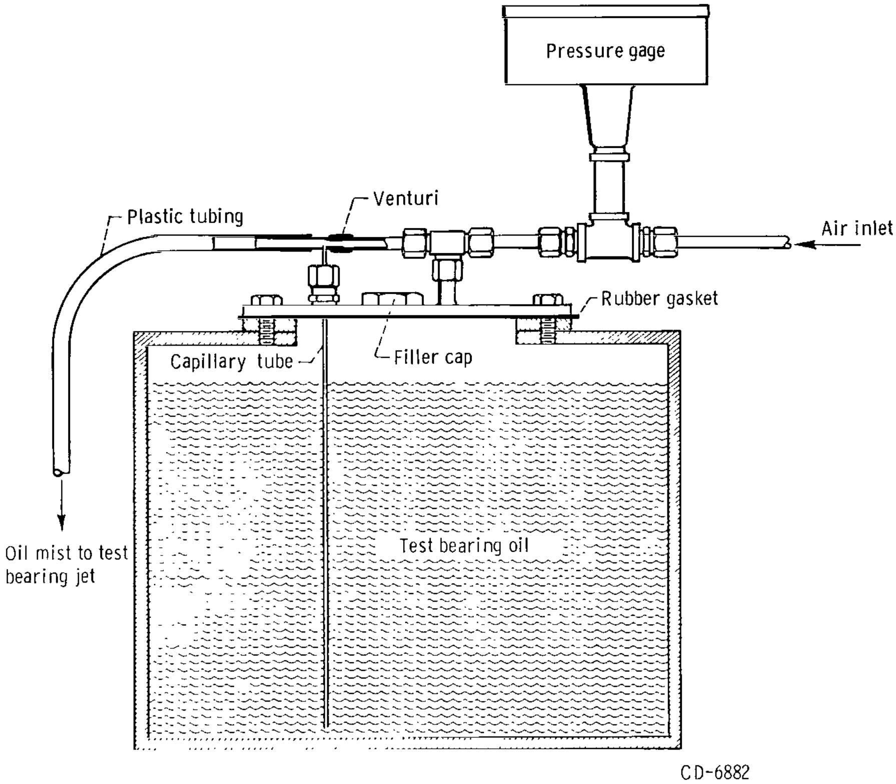
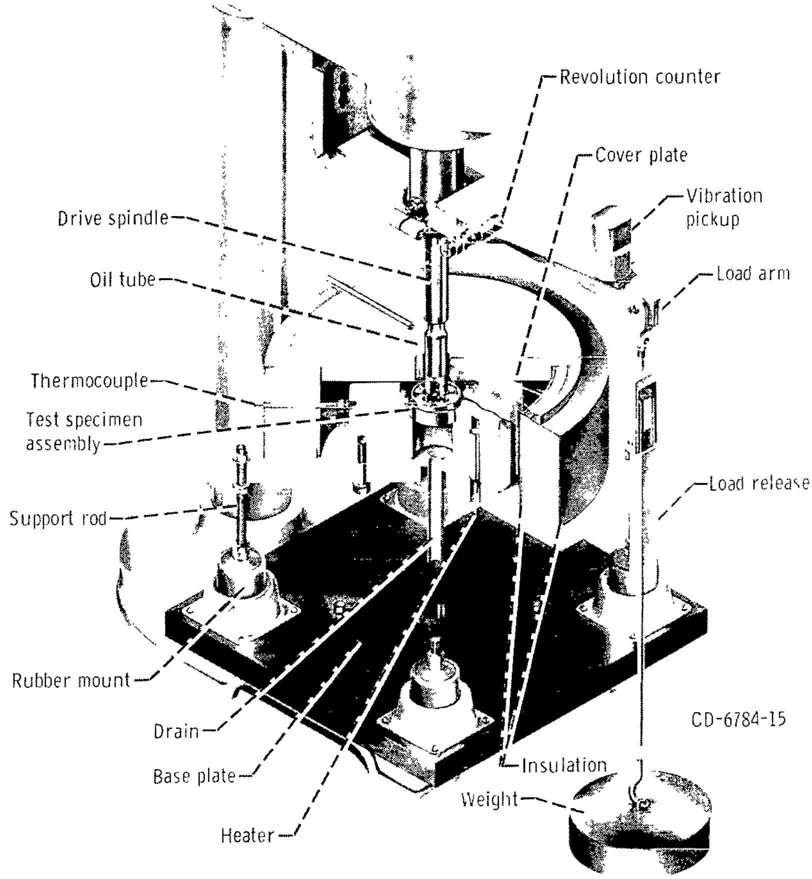


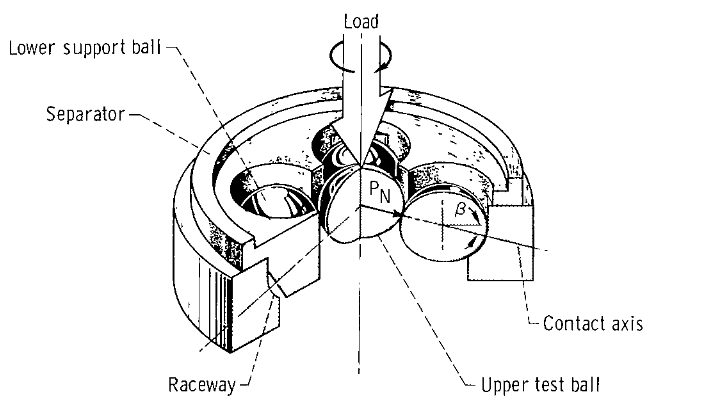

Figure 2. - Air-oil-mist test bearing lubrication system.

## Five-Ball Fatigue Tester

The NASA five-ball fatigue tester was used for the hollow ball fatigue tests. The apparatus (fig. 3) is identical to that used in reference 9 and is described in detail in reference 11. This fatigue tester consists essentially of an upper test ball pyramided on four lower support balls that are positioned by a separator and are free to rotate in an angular-contact raceway. The load is applied to the upper test ball through a vertical shaft that drives the ball assembly. The operation of the test ball and raceway is analogous to that of the inner and outer races of a bearing, respectively. The separator and the lower support balls function like the cage and the balls in a bearing, respectively.




(a) Cutaway view of five-ball fatigue tester.




(b) Test specimen assembly.

Figure 3. - Five-ball fatigue tester.

## Test Bearings and Hollow Balls

The test bearing specifications are listed in table I. Two series (115 and 215) of 75-millimeter-bore deep-groove ball bearings were used with both solid and hollow balls. The series 115 bearings used 1/2-inch- (12.7-mm-) diameter balls, and the series 215 bearings used 11/16-inch- (17.5-mm-) diameter balls. The test bearings are shown in figure 4.

The 1/2-inch- (12.7-mm-) diameter hollow balls had a wall thickness of 0.100 inch (2.5 mm). The ratio of the outside to the inside diameter was 1.67, and the weight reduction was 21.7 percent from that of a solid ball. The hollow 1/2-inch- (12.7-mm-) diameter balls were made from SAE 52100 consumable-electrode vacuum-melted (CVM) steel.

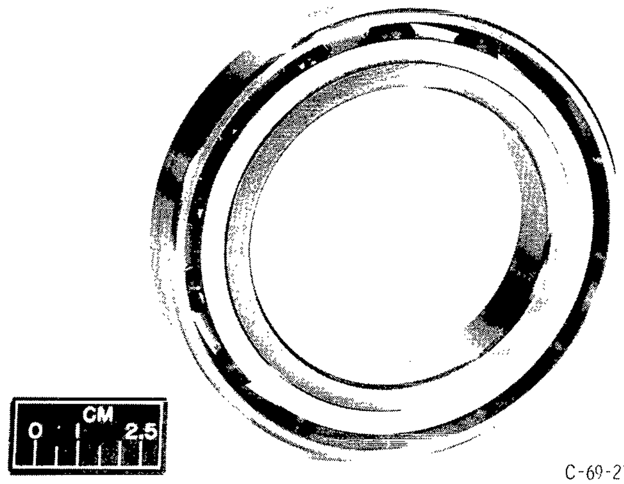
The 11/16-inch- (17.5-mm-) diameter hollow balls had a wall thickness of 0.060 inch (1.5 mm). This is an outside- to inside-diameter ratio of 1.2 with a weight reduction of 56.5 percent from that of a solid ball. The 11/16-inch (17.5-mm) hollow balls were fabricated from AISI M-50 CVM steel.

TABLE I. - DEEP-GROOVE 75-MILLIMETER-BORE BEARING SPECIFICATIONS

| Bearing[a] | | | | Ball | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Series | Outer diameter, mm | Radial clearance | | Number per bearing | Outside diameter | | Wall thickness | | Material[b] | Weight reduction, percent | Ratio of outside to inside diameter |
| | | in. | mm | | in. | mm | in. | mm | | | |
| 115 | 115 | 0.0027 | 0.069 | 14 | 0.500 | 12.7 | Solid | Solid | SAE 52100 | ---- | ---- |
| 115 | 115 | .0027 | .069 | 14 | .500 | 12.7 | 0.100 | 2.5 | SAE 52100 | 21.7 | 1.67 |
| 215 | 130 | .0020 | .051 | 11 | .6875 | 17.5 | Solid | Solid | AISI M-2 | ---- | ---- |
| 215 | 130 | .0020 | .051 | 11 | .6875 | 17.5 | .060 | 1.5 | AISI M-50 | 56.5 | 1.21 |

| Race | | | Retainer | | |
|---|---|---|---|---|---|
| Curvature | | Material[b] | Design | Locating surface | Material |
| Inner | Outer | | | | |
| 0.54 | 0.54 | SAE 52100 | One piece, machined | Inner race | Silver-plated bronze |
| .54 | .54 | | | | |
| .53 | .53 | AISI M-2 | Two piece, machined, riveted | Outer race | Annealed AISI M-2 |
| .53 | .53 | | | | |

[a]Grade ABEC-5.
[b]Consumable-electrode vacuum-melted.

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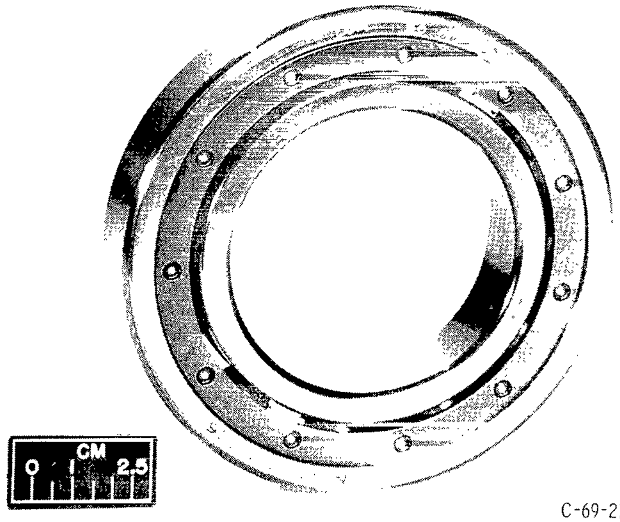
(a) Series 115: separable at outer race; one-piece cage construction; cage material, silver-plated iron-silicon-bronze.

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(b) Series 215: deep groove; two-piece machined cage construction; cage material, AISI M-2 tool steel.

Figure 4. - Test bearings.

TABLE II. - CHEMICAL ANALYSIS OF HOLLOW BALL MATERIALS

| Material[a] | C | Si | Mn | S | P | W | Cr | V | Mo | Co | Ni | Cu |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Analysis, wt.% | | | | | | | | | | | |
| SAE 52100 | 1.03 | 0.26 | 0.34 | 0.005 | 0.010 | ---- | 1.53 | ---- | 0.03 | ---- | 0.05 | 0.08 |
| AISI M-50 | .79 | .22 | .23 | .004 | .008 | 0.02 | 4.05 | 1.04 | 4.20 | 0.02 | .04 | .02 |

[a]Consumable-electrode vacuum-melted.

TABLE III. - HOLLOW BALL MATERIAL HEAT TREATMENT

[Consumable-electrode vacuum-melting process.]

| Material | Anneal after electron-beam welding | Austenitize | Quench | Temper | | | Nominal Rockwell C hardness |
|---|---|---|---|---|---|---|---|
| | | | | First | Second | Third | |
| SAE 52100 | Heat to 1475° F (1079 K)<br>Hold for 7 hours[a]<br>Furnace cool to 300° F (422 K) at 20° F per hour (6.5 K/hr)<br>Air cool to room temperature | 1550° F (1118 K) for 12 minutes | Oil to room temperature | 1020° F (822 K) for 180 minutes | 1045° (836 K) for 120 minutes[c] | 1050° F (838 K) for 120 minutes | 65 |
| AISI M-50 | Heat to 1550° F (1118 K)<br>Hold for 2 hours[b]<br>Furnace cool to 1050° F (838 K) at 50° F per hour (10 K/hr)<br>Air cool to room temperature | 2065° F (1403 K) for 6 minutes | Molten salt to 1050° F (838 K)<br>Air cool to room temperature | 275° F (408 K) for 60 minutes | ---------------- | ---------------- | 63.5 |

[a]In spent cast iron chips.
[b]In stainless-steel bag.
[c]Subzero cool to -150° F (172 K) for 2 hr.

A chemical analysis of the material from which each size hollow ball was made is given in table II. The heat treatment schedule for each material is given in table III.

# PROCEDURE

## Bearing Tests

Each test bearing was run in for 1 hour with a 250-pound (1100-N) thrust load, a shaft speed of 6000 rpm, and an air-oil-mist flow rate of about 0.010 pound per minute ($0.8\times10^{-4}$ kg/sec). After run-in, the air-oil-mist flow rate was increased to 0.037 pound per minute ($2.8\times10^{-4}$ kg/sec), the thrust load was set at 500 pounds (2200 N), and the shaft speed was increased to 10 000 rpm. Each bearing was operated at this initial condition until temperature equilibrium was achieved. Equilibrium was assumed for each data point when the bearing outer-race temperature reading had not changed more than 1° F (2 K) over a 10-minute interval.

After the initial data point was taken at a speed of 10 000 rpm and a thrust load of 500 pounds (2200 N), the shaft speed was increased in increments of 2000 rpm while the

TABLE IV. - LUBRICANT FLOW RATE

| Shaft speed, rpm | Thrust load, lb (N) | | | |
|---|---|---|---|---|
| | 500 (2200) | | 1000 (4400) | |
| | Oil flow rate[a] | | | |
| | lb/min | kg/sec | lb/min | kg/sec |
| 10 000 | 0.037 | 0.00028 | 0.037 | 0.00028 |
| 12 000 | .037 | .00028 | .037 | .00028 |
| 14 000 | .037 | .00028 | .037 | .00028 |
| 16 000 | .046 | .00035 | .062 | .00047 |
| 18 000 | .062 | .00047 | .073 | .00055 |
| 20 000 | .073 | .00055 | ----- | ------- |
| 22 000 | .073 | .00055 | ----- | ------- |
| 24 000 | .073 | .00055 | ----- | ------- |

[a]Oil introduced to bearing as an air-oil mist.

load was maintained constant. The lubricant flow rate was increased with the shaft speed, as shown in table IV. The technique described in reference 12 was used to determine the flow rates required. Outer-race temperature and bearing torque were recorded for each shaft speed. Data points were taken until outer-race temperature equilibrium could not be attained. This procedure was repeated for the 1000-pound (4400-N) thrust load tests. The highest maximum Hertz stress was about 280 000 psi ($1.9\times10^9$ $N/m^2$) at the ball-race contacts for the series 115 bearings and about 250 000 psi ($1.7\times10^9$ $N/m^2$) for the series 215 bearings.

## Fatigue Tests

Fatigue tests were conducted in the five-ball fatigue tester at a maximum Hertz stress of 700 000 psi ($4.82\times10^9$ $N/m^2$). The drive-shaft speed was 10 600 rpm, and the contact angle (indicated by $\beta$ in fig. 3) was $34.5^\circ$. Tests were run continuously for 300 hours, or until failure of the upper test ball or a lower support ball occurred. The outer-race temperature stabilized at about $145^\circ$ F (336 K) with no heat added. The 11/16-inch- (17.5-mm-) diameter hollow balls were run with 1/2-inch (12.7-mm) solid AISI M-50 lower support balls with a Rockwell C hardness of 64 to 64.5. The lubricant was identical to the super-refined naphthenic mineral oil used in the bearing tests.

## Post-Test Inspection

After the testing, the bearings were disassembled, cleaned, and inspected for damage.

After the fatigue tests, the hollow balls with failures were cleaned and examined. The balls were electropolished to determine the location of the weld, the orientation of the weld relative to the ball track, and the proximity of the spall to the weld area.

Some balls were sectioned through and parallel to the track. The sections were polished and photomicrographs were taken.

# RESULTS AND DISCUSSION

## Bearing Tests

Two series (115 and 215) of 75-millimeter-bore ball bearings were tested with solid and hollow rolling elements to determine their operating characteristics. The results of tests with the series 115 bearings are shown in figure 5. Very little difference existed

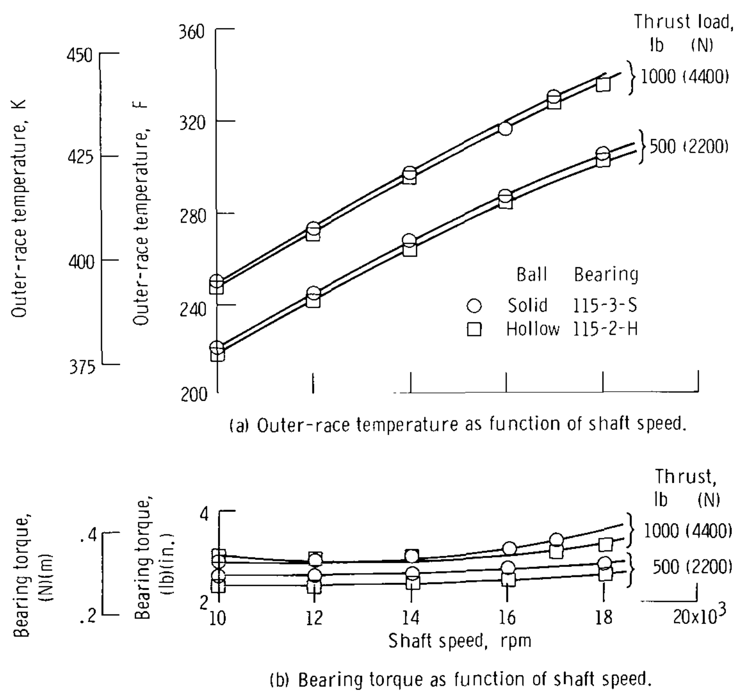
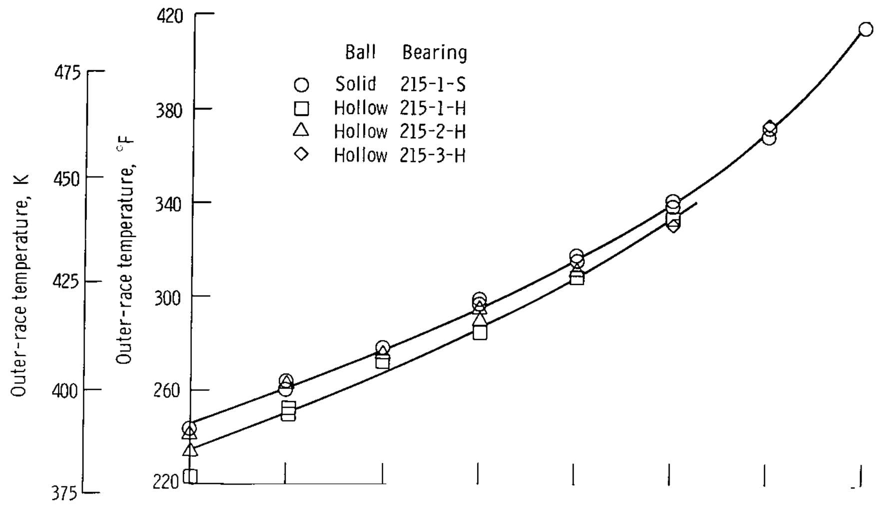
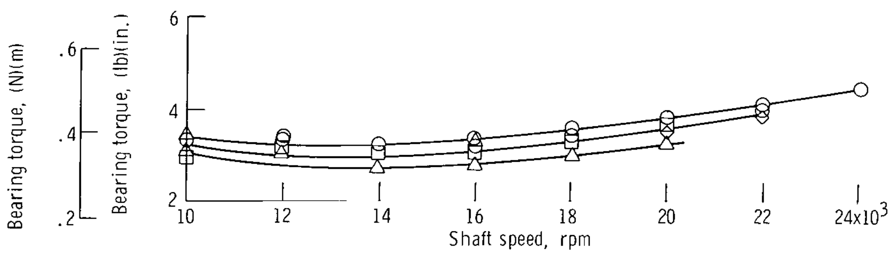



Figure 5. - Comparison of performance data of series 115 75-millimeter bearings using both solid and hollow balls. Ball diameter, 1/2 inch (12.7 mm); ball material, SAE 52100 steel; bearing specifications given in table I.




(a) Outer-race temperature as function of shaft speed.




(b) Bearing torque as function of shaft speed.

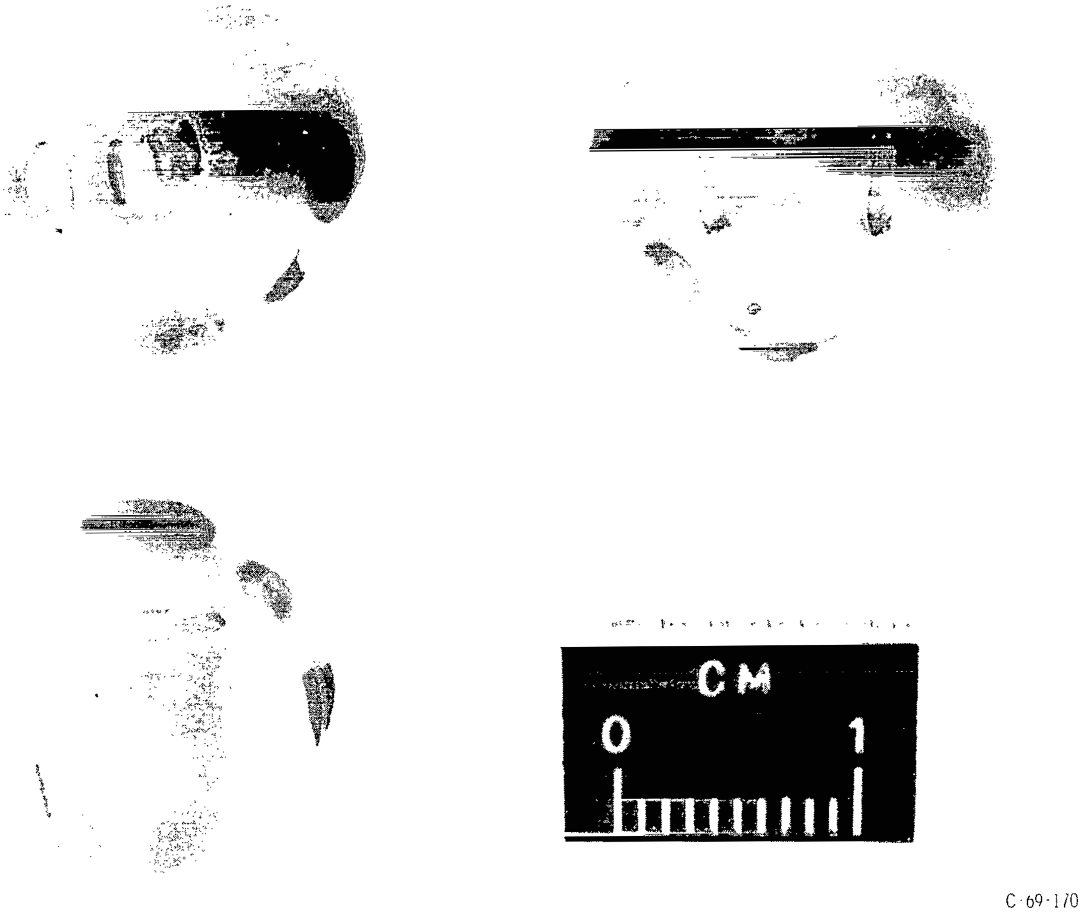
Figure 6. - Comparison of performance data of series 215 75-millimeter bearings using both solid and hollow balls. Ball diameter, 11/16 inch (17.5 mm); ball material, AISI M-2 for solid balls and AISI M-50 for hollow balls; thrust load, 500 pounds (2200 N); bearing specifications given in table I.

in either the outer-race temperature (fig. 5(a)) or the total bearing torque (fig. 5(b)) between the bearing with the solid balls and the bearing with the hollow balls over the range of shaft speeds investigated.

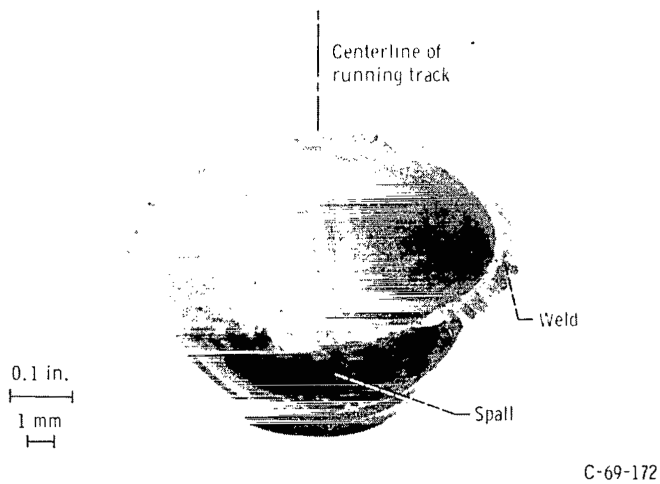
The results of tests with the series 215 bearings are shown in figure 6. The curves of outer-race temperature (fig. 6(a)) and bearing torque (fig. 6(b)) against speed show that the hollow ball bearings had slightly lower values than the solid ball bearing for the range of shaft speeds investigated.

## Post-Test Inspection of Bearings

Visual inspection of the series 115 bearings revealed that the solid and hollow balls, races, and retainers were in very good condition with no apparent wear.




(a) Balls from bearing 215-1-H showing external spalls.




(b) Ball from bearing 215-3-H showing complete fracture.

Figure 7. - Typically damaged hollow balls from series 215 bearings.

Inspection of the series 215 bearings, however, revealed extensive damage to the hollow balls. A photograph showing typical damage experienced by the balls is presented in figure 7. The ball damage ranged from spalling (fig. 7(a)) to complete fracture (fig. 7(b)). The spalls were all on the running track. The ball in figure 7(b) fractured into two halves at the weld. The track was perpendicular to the weld, and a spall was right at the weld. The total running times and damage to the hollow balls of each bearing are summarized in table V.

TABLE V. - SUMMARY OF BALL DAMAGE OF SERIES 215 BEARINGS

| Bearing | Running time, hr | Condition of balls |
|---|---|---|
| 215-1-H | [a]13.6 | Nine spalled balls |
| 215-2-H | 8.6 | One spalled ball that broke when bearing was dissembled |
| 215-3-H | 4.1 | One fractured ball, four spalled balls, and one ball with surface crack |

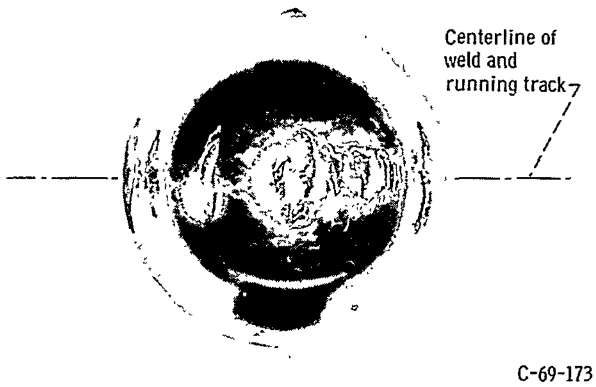
[a]Bearing apparently ran for about 3 hr after first spall occurred.

One ball from bearing 215-1-H had 14 spalls around its circumference (fig. 8(a)). After this ball was sectioned and examined, it was determined that the ball track coincided with the plane of the weld, as shown in figures 8(a) and (b). Further examination of this ball revealed many cracks on the internal surface, across the weld, as shown in figure 8(c). This ball was then sectioned parallel to and through the track, and the specimen was polished. A composite of several photomicrographs, shown in figure 8(d), reveals a pattern of cracks extending from the inside of the ball to the outside surface spalls.

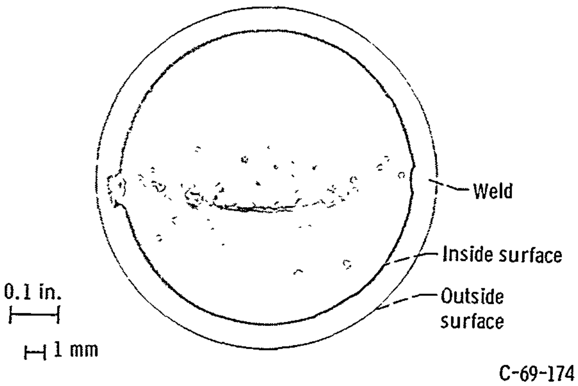
Other balls that were damaged were electropolished to determine the orientation of the weld relative to the running track. All these tracks were at angles from $45^{\circ}$ to $90^{\circ}$ to the weld. Every ball that was sectioned parallel to and through the track revealed cracks between the weld area and the outside surface, and example of which is shown in figure 9.

It was not apparent from these photographs whether the cracks had originated at the outside surface and propagated inward or had originated at the inside surface and propagated outward through the wall, resulting in a surface spall. However, several balls that did not show any surface distress were sectioned parallel to and through the ball track. In each of these balls, cracks were revealed that apparently had originated at the inside surface in the weld area, as shown in figure 10.

Several observations on the integrity of the weld were made. Microhardness readings were taken on a sectioned ball in the weld zone and in the parent material. A vari-

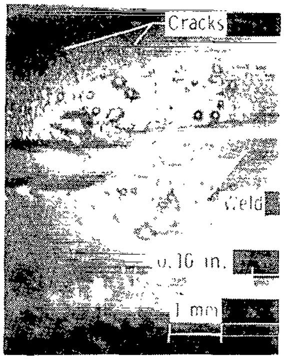



(a) View of external surface showing multiple spalls.

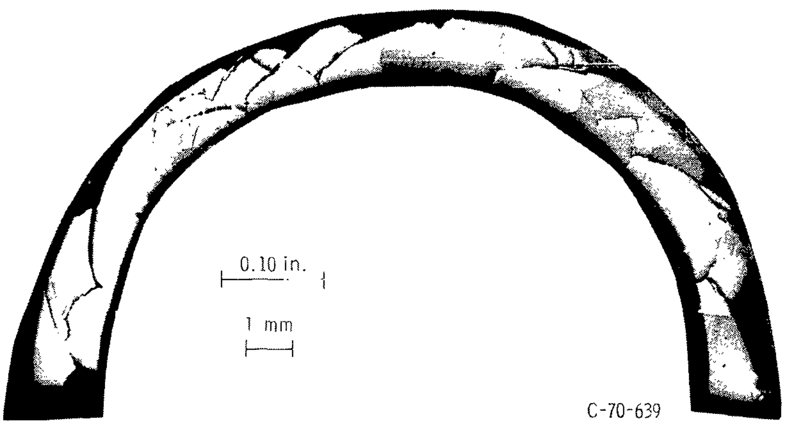



(b) View of internal surface showing weld.

Figure 8. - Ball from bearing 215-1-H with running track directly on weld area.




(c) Closeup view of weld bead showing cracks on internal surface, across weld.




(d) Composite of photomicrographs of ball sectioned parallel to and through both running track and weld.

Figure 8. - Concluded.

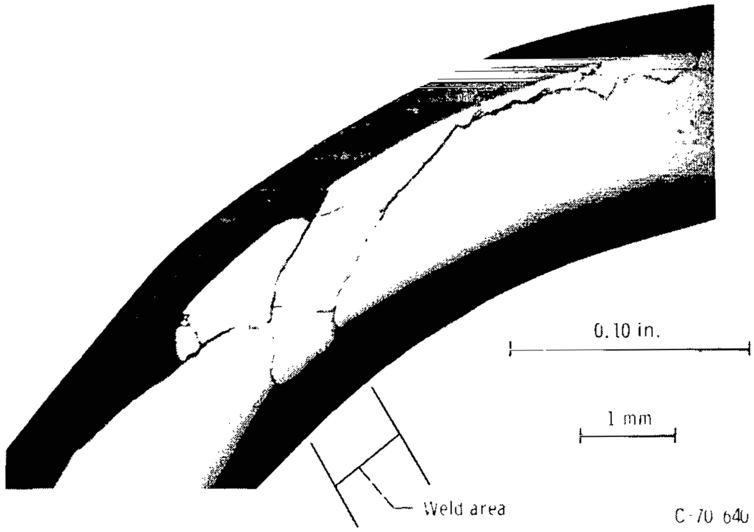



Figure 9. - Sectional view showing typical cracks between weld area and outside surface of ball. Section taken parallel to and through running track.

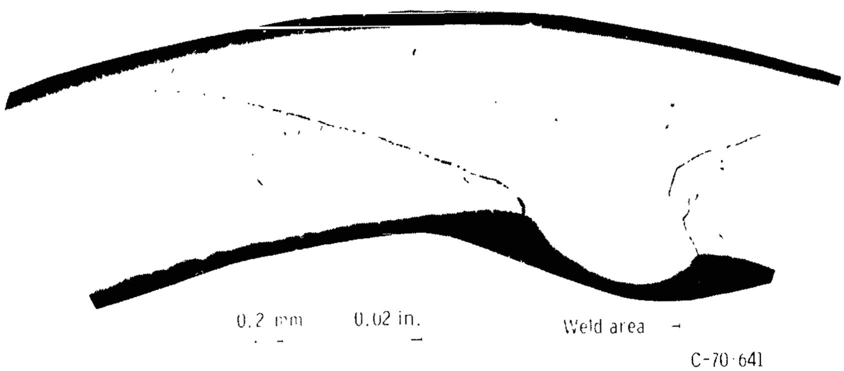



Figure 10. - Section of typical ball (from bearing 215-1-H) on which no surface spalls were evident. Cracks shown originated at weld area. Section taken parallel to and through running track.

ation in Rockwell C hardness of about one point was observed. A very slight difference in microstructure was observed, but this difference was not considered significant.

The electron-beam weld appeared to be of good quality and proper technique. The bead on the inside surface of the balls, however, varied in thickness around the circumference. There was considerable spatter of material on the inside surface of the ball. In some sectioned specimens, the wall thickness was slightly less adjacent to the weld than in the remainder of the wall.

Two hollow balls that had not been run in a bearing were examined by X-ray techniques, and no cracks were observed. Three hollow balls that had been run in a bearing but had no apparent surface defects were then examined by X-ray, and cracks in the weld area were observed in all three.

From the aforementioned examination, it was concluded that the hollow balls from the series 215 bearings had probably failed in flexure because of a stress concentration in the region of the weld. This stress concentration appeared in nearly all cases to be caused by the notch formed at the edge of the weld bead.

## Fatigue Tests

The results of fatigue tests with 1/2-inch- (1.27-mm) diameter balls in the NASA five-ball fatigue tester are presented in reference 9. The fatigue lives of those hollow balls (0.100-in. or 2.54-mm wall thickness) compared favorably with solid balls of the same material. It was concluded that the weld zone material on the hollow balls was not significantly weaker in fatigue than the solid parent material.

The results of fatigue tests with the 11/16-inch- (17.5-mm-) diameter hollow balls (0.060-in. or 1.5-mm wall thickness) are presented in figure 11. The balls were run in the five-ball fatigue tester under conditions similar to those of reference 9. Figure 11 is a plot of the log-log of the reciprocal of the probability of survival as a function of the log

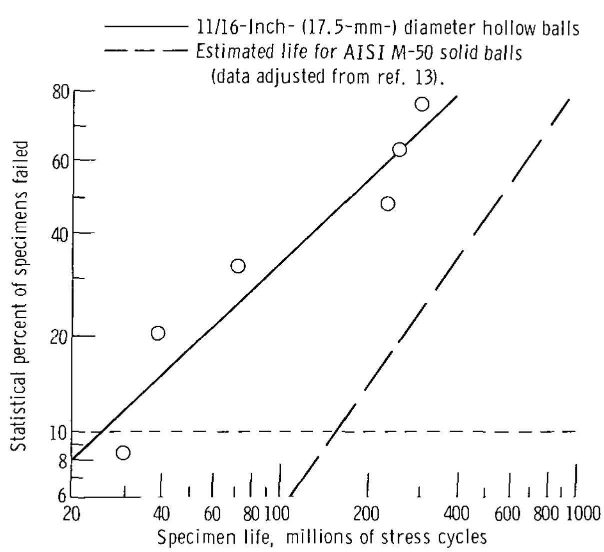



Figure 11. - Rolling-element fatigue life of AISI M-50 steel hollow and solid upper balls in five-ball fatigue tester. Maximum Hertz stress, 700 000 psi ($4.82\times10^{9}$ N/m$^2$); speed, 10 600 rpm; lubricant, naphthenic mineral oil; no heat added.

of the stress cycles to failure (Weibull coordinates). For convenience, the ordinate is the statistical percent of specimens failed. For comparison, data for M-50 steel solid balls from reference 13 were adjusted for differences in stressed volume and contact stress and were plotted in figure 11. The fatigue lives are obviously much lower for the hollow balls than for the solid balls.

## Post-Test Inspection of Fatigue Balls

The 11/16-inch- (17.5-mm-) diameter hollow balls that had been tested on the five-ball fatigue tester were electropolished to determine the orientation of the weld. On each of the six failed balls, the running track crossed the weld. The fatigue spall on five of these six balls was adjacent to or directly on the weld. The spall on the other ball was clearly away from the weld area. On the one ball that had no fatigue spall after 530 million stress cycles (300 hr), the running track did not cross the weld.

Four of the failed balls with the spall adjacent to the weld were sectioned parallel to and through the running track. In each ball, a crack connected the outside surface spall to the inside surface and passed through the weld area. A typical crack is shown in figure 12.

From the width, shape, and pattern of these cracks, it was concluded that they initiated at the inner surface of the ball and propagated to the outer surface, resulting in a spall that resembled those of classical subsurface rolling-element fatigue. It was also concluded that these hollow balls had probably failed in flexure because of a stress con-

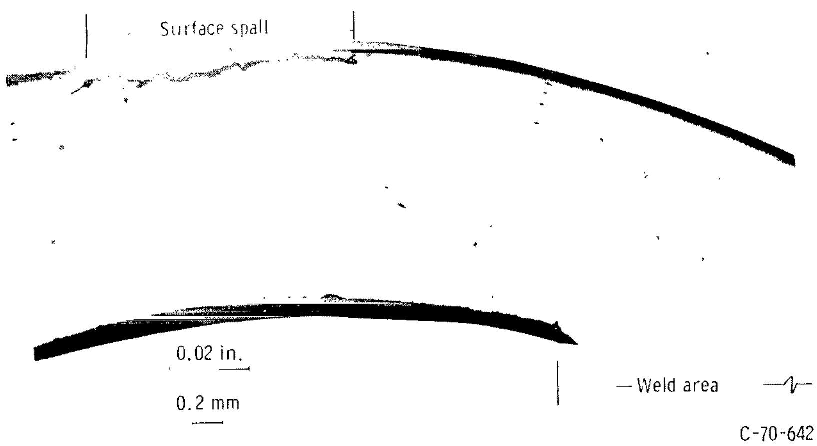



Figure 12. - Typical section through junction of weld and running track showing crack connecting inside of wall with spall on outer surface. AISI M-50 steel 11/16-inch- (17.5-mm-) diameter hollow ball (upper test ball in five-ball fatigue tester). Maximum Hertz stress, 700 000 psi ($4.82 \times 10^9$ N/m$^2$); etchant, Picral-hydrochloric acid.

centration in the region of the weld, like the balls in the series 215 bearings. Again, the stress concentration appeared to be at the notch formed at the edge of the weld at the ball inner diameter.

The ball with the spall away from the weld area was also sectioned through the spall, parallel to the running track. No cracks were seen connecting the spall and the inner surface of the ball. This failure was apparently not a result of flexure of the wall and most probably was classical subsurface fatigue.

## CONCLUDING REMARKS

The results of reference 9 showed that 1/2-inch- (12.7-mm-) diameter SAE 52100 steel hollow balls, with an outside- to inside-diameter ratio of 1.6, compared favorably with solid balls in fatigue tests. However, the 11/16-inch- (17.5-mm-) diameter M-50 steel hollow balls, with a diameter ratio of 1.2 reported herein, did not compare well with solid balls in fatigue tests. The fact that the balls were of different material and size should not have made any significant difference in the fatigue comparisons. These results suggest that the diameter ratio of 1.2 results in a wall that is too flexible for use in a bearing ball.

The series 115 hollow ball bearings (diameter ratio = 1.6) did not have any ball failures but showed no marked improvement in torque or temperature over the solid ball bearings over the range of shaft speeds tested. Although the series 215 hollow ball bearings (diameter ratio = 1.2) had ball failures, they did show some improvement in torque and temperature over the solid ball bearings. However, a combination of the air-oil-mist lubrication system and the temperature limitation of the SAE 52100 steel limited the testing of the series 115 bearings to moderate speeds. This does not preclude the possibility that some improvement in torque or temperature would be noted if the bearing could have been operated at higher shaft speeds.

A number of problems are inherent in the manufacture of hollow balls that are to be used in bearings. Two significant problems are the difficulty in (1) maintaining uniform wall thickness and (2) controlling the weld penetration around the periphery of the ball. Therefore, the balls will almost always have an unbalance and/or a slightly different stiffness at the weld. Under dynamic conditions, these factors could adversely affect the life of the ball.

The hollow ball failures reported herein originated at a stress concentration at the ball inner diameter. It may therefore be concluded that, where stress raisers exist, early failures will result when the wall of a hollow ball is sufficiently thin and flexure becomes significant.

## SUMMARY OF RESULTS

An experimental investigation was conducted to determine the operating characteristics of ball bearings using hollow balls as the rolling elements. Similar bearings using solid balls were also tested and the data compared. Two series (115 and 215) of 75-millimeter-bore deep-groove ball bearings were operated at shaft speeds up to 24 000 rpm ($1.8\times10^6$ DN) with thrust loads of 500 and 1000 pounds (2200 and 4400 N). The series 115 bearings used 1/2-inch- (12.7-mm-) diameter balls with a 0.100-inch (2.54-mm) wall thickness, and the series 215 bearings used 11/16-inch- (17.5-mm-) diameter balls with a 0.060-inch (1.5-mm) wall thickness. The torque and outer-race temperature data of the bearings with the hollow balls were compared with data of similar bearings with solid balls. Tests were conducted in the NASA five-ball fatigue tester to determine the life of the 11/16-inch- (17.5-mm-) diameter hollow balls. The following results were obtained:

1. The 11/16-inch- (17.5-mm-) diameter hollow balls failed by flexure fatigue. Cracks initiated at the inner diameter at a stress concentration in the weld area and propagated through the wall, terminating in a spall on the outer surface.

2. The rolling-element fatigue life of the 11/16-inch- (17.5-mm-) diameter hollow balls, determined at a maximum Hertz stress of 700 000 psi ($4.82\times10^9$ N/m$^2$), was significantly less than lives predicted from previously run AISI M-50 solid steel balls. The hollow balls from the fatigue tests appeared to have failed in flexure, like the hollow balls from the series 215 bearings.

3. The torques and outer-race temperatures of the series 115 hollow ball bearings operating at speeds up to 18 000 rpm ($1.35\times10^6$ DN) were not significantly different from those of the series 115 solid ball bearings.

4. The torque and outer-race temperature of the series 215 hollow ball bearings were slightly lower than the values for the series 215 solid ball bearings at corresponding test conditions.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, March 3, 1970,
126-15.

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